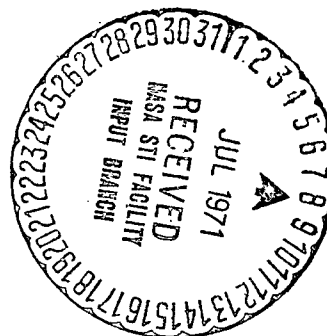


NAS3-11826

THE DYNAMIC CHARACTERISTICS OF A TURBO-ROTOR SIMULATOR  
SUPPORTED ON GAS-LUBRICATED FOIL BEARINGS

Part II: Operation with Heating and Thermal Gradients

by  
L. Licht\*



N72-12411 (NASA-CR-124621) THE DYNAMIC  
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(CATEGORY)

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ABSTRACT

A high-speed rotor, supported by gas-lubricated foil bearings, is free from self-excited whirl and displays no loss of load capacity when vibrated at frequency equal half the rotational speed [1].\* It is demonstrated here that in addition to tolerance of geometrical imperfections, misalignment and foreign particles [3, 4], the foil bearing performs well at elevated temperatures and accommodates appreciable temperature gradients. The foil bearing is endowed with superior wipe-wear characteristics and the flexibility of the foil accounts not only for the stability of the foil bearing, but also for its forgiveness with respect to distortion, contamination and contact.

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\*Numbers in brackets designate References at end of paper.

## INTRODUCTION

The emphasis in the first part of this investigation [ 1 ] was concentrated mainly on the response of a high speed, foil-bearing supported simulator of a turbine-compressor-alternator unit to various types of excitation. In this paper attention is focussed on the effects of nonuniform heating and the performance of the foil bearing in the presence of appreciable temperature gradients.

In the case of gas bearings operated at elevated temperatures, serious difficulties are posed by thermal gradients and the accompanying thermal distortions. Not only is the performance of rigid gas bearings extremely sensitive to clearance and the shape of the fluid film, but the mere existence of separation between journal and bearing surfaces may be threatened by differential expansion. The point is that gas-bearing clearances are relatively small, small in comparison with changes in journal and bearing diameters at elevated temperatures. The control of this difference in dimensions is always difficult. A gas bearing which provides a whirl-free and reliable rotor support in the absence of heating, and accommodates resonant amplitudes of motion within the nominal clearance circle, may pose a host of problems in a high-speed turbo-machine operating at elevated temperatures.

The foil bearings, which were shown to be stable and free from the "half-frequency sensitivity," have also proven their distortion-accommodating and superior wipe-wear characteristics in the absence of heating [2,3,4]. Operation in the presence of appreciable thermal gradients has not previously been attempted. To simulate the heat flow from the turbine to the adjacent journal and foil, a spirally wound heating element was placed at a distance approximately 0.1 inch opposite the rotating end-disc.\* Both heater and the turbine-simulating disc were placed in an insulating

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\*The reader is referred to Fig. 3, Part I [ 1 ], which shows details of construction of the experimental apparatus.

enclosure, consisting of a capped, cylindrical side-wall and a base with a 2 inch diameter opening. The experimental apparatus in the vertical attitude is illustrated in Fig. 1, with details of heater and thermocouple arrangement shown in Fig. 2.

#### DETERMINATION OF FOIL AND JOURNAL SURFACE - TEMPERATURES

The base of the heater assembly was attached to the outboard set of foil guides adjacent to the heating enclosure. The thermocouples numbered 1 through 6, six holders, and the location of junctions along the bisector of adjacent foil segments are clearly discernible in the view of Fig. 2.\* The thermocouples A and B were located within the heating enclosure. The shielded junction of thermocouple A was situated approximately 0.2 inch above the center of the heated disc and its temperature could be considered of the same order as that of the air in the heater-disc interspace. The junction of the fine-wire thermocouple B was located approximately 0.2 inch above the journal shoulder and approximately 0.003 to 0.005 inch from the short cylindrical section connecting the heated disc and the journal. Its maximum temperature was probably lower, but of the same order as the maximum surface temperature at the journal end. The output of thermocouple C was representative of the ambient air temperature in the vicinity of the test apparatus. The reader may note that the inboard (lower) foil in Fig. 2 was not swept directly by the cool turbine-air discharge, which was deflected around the inside radius of the rotor flange, just above the nozzle ring. The outboard (upper) foil was shielded from convective air currents by the foil support plate and by a screen surrounding the foil and foil guides (placed between the heater base and the support plate and not shown in Fig. 2). The heater and thermocouple arrangement is also illustrated schematically in Fig. 3 and Fig. 5.

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\*Chromel-Alumel, 0.003 in diameter wires. Welded junctions and adjacent wire attached to foil under microscope with small amounts of silicone adhesive.

In addition to thermocouples, temperature-sensitive coatings\* were applied locally at various foil locations. The latter, though available at discrete intervals of  $25^{\circ}\text{F}$  only, had a specified accuracy of melting point of 1% and furnished useful means of estimating differences between foil and thermocouple temperatures. It appears that this difference, though relatively insignificant at temperatures the order of  $200^{\circ}\text{F}$ , showed the thermocouple readings to be lower by 10% to 14% at temperatures the order of  $500^{\circ}\text{F}$ . The foregoing should be taken into consideration in the interpretation of thermocouple data with respect to foil temperatures in Figs. 3, 4, 5 and 6.

Thin layers of temperature-sensitive coatings were also applied close to the outer journal edge, and to the journal surface just outside the foil edge adjacent to thermocouple No. 6. Parts of these coatings adhered with sufficient tenacity to withstand the centrifugal forces and finally melted, leaving tell-tale streaks and small traces of crystals as proof of melting at specific locations.

### HISTORIES OF HEATING CYCLES

Histories of heating cycles in the vertical attitude were obtained at rated speed  $N = 36,000\text{ RPM}$ . In addition to thermocouple readings and observation of melting points of coatings on the foil surface, recorded also was the approximate heater power input and the turbine nozzle supply pressure required to maintain rated speed. The history of the first heating cycle, of approximately 2 hours' duration, is presented graphically in Fig. 3. The foils used in this experiment were 0.001 inch thick molybdenum (coefficient of thermal expansion  $2.7 \times 10^{-6}\text{ in/in/}^{\circ}\text{F}$  - as compared with  $5.6 \times 10^{-6}\text{ in/in/}^{\circ}\text{F}$  for AISI 440-C, the rotor material - and modulus of elasticity approximately  $47 \times 10^6\text{ psi}$ ).

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\*Tempilaq (trade name).

After a 30 minute warm-up without heat input, the rate of power dissipation in the heater was increased rather rapidly. Because of thermal lag and delay in power-supply cutoff, the temperatures attained in the vicinity of the journal and foil edges adjacent to the heater exceeded greatly the intended magnitudes, reaching approximately 800<sup>o</sup>F at the foil surface and, possibly, even higher temperature at corresponding points on the journal. Closure of the gap required increased power to drive the rotor, as reflected by the increase of supply pressure at the turbine nozzles from 51 psig to 68 psig. The rotor, nevertheless, continued to operate smoothly at 600 RPS and the orbital motion was negligibly small.

Since rising foil temperatures, despite first diminished and then discontinued heat input, indicated a very large reduction in gap width, and possibly some metal contact, the rotor was coasted down after approximately 2 hours of operation (Fig. 3). Deceleration was very rapid and the usual increase in gap width, due to pressurization may have been offset by local contraction of the foil and contact along a row of cooling orifice jets.

Despite rather excessive heating in the course of the first run, there were two surprising and encouraging aspects of the experiment. The first was that the rotor remained operational and that it has been possible to maintain high-speed rotation under such adverse thermal conditions. The second, and even more surprising experience, was that the rotor, after a short period of cooling, could be restarted and did perform faultlessly throughout the entire experimental speed range. We anticipated also that foil and journal damage might be quite extensive. It now appears, as will be shown in the following section, that these fears were not justified.

The temperature profiles along bisectors of the two adjacent foil-bearing segments, recorded at various times in the course of the first run, are shown graphically in Fig. 4. Disregarding the 125-minute curve as far too extreme for conditions to be anticipated in an actual turbomachine, we



note that the foil gradients of the 105-minute and 85-minute profiles are the order of  $135^{\circ}\text{F}/\text{in}$  and  $125^{\circ}\text{F}/\text{in}$  respectively.\*

Prior to the second experimental run with heat input, the 0.001 inch thick molybdenum foil adjacent to the heater was replaced by an Inconel-600 foil of equal thickness. The objective of this change was to obtain a better matching of coefficients of thermal expansion to compensate for dimensional changes of rotor, foil and foil supports. (The coefficients of thermal expansion of metals involved in this case were 2.7, 5.6, 6.5 and  $7.4 \mu\text{in}/\text{in}/^{\circ}\text{F}$  for molybdenum, AISI 440-C, AISI 416 and Inconel-600 respectively). Some relief was also to be expected as a result of lower extensional rigidity of Inconel-600, due to its lower elastic modulus. The duration of the run was approximately 8 hours, at rated speed  $N = 36,000$  RPM.

The history of this heating cycle and corresponding foil-temperature profiles are illustrated graphically in Figs. 5 and 6. In this experimental run, successive increments of power input to the heater were smaller and spread over longer time periods. The rotor was first accelerated to rated speed and operated without heat input for 90 minutes, allowing the foils to reach equilibrium temperatures. It can be seen that the energy dissipated in shearing the air film produced a sensibly uniform temperature the order  $180^{\circ}\text{F}$  along the shielded, outboard foil. The inboard foil, exposed to convective air currents, was cooler, with temperature dropping in the direction of the air turbine. The latter furnished an efficient heat sink in the midplane of the rotor.

The temperature attained by the outer edge of the journal was between  $575^{\circ}$  and  $600^{\circ}\text{F}$ . The maximum temperature attained at the other journal extremity, approximately 0.125 inch from the foil edge, was less than  $125^{\circ}\text{F}$ . The maximum temperature of the foil adjacent to the heater was between  $525^{\circ}$  and  $550^{\circ}\text{F}$ , dropping by approximately  $100^{\circ}\text{F}$  across the width of the foil. The corresponding differential across the inboard foil

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\*The corresponding axial temperature gradients on the journal surface would be at least as steep, or steeper.

was also the order of  $100^{\circ}\text{F}$ , from a maximum between  $350^{\circ}$  and  $325^{\circ}\text{F}$ . The foregoing values were established on the basis of melting points of temperature-sensitive coatings (See Fig. 6).

The highest bearing temperatures attained in the course of this run were approximately 30% lower than the maxima reached during the first run, but generally higher than values to be encountered in an actual machine. The supply pressure at the nozzles (Fig. 5) was increased from 51 to 57 psig only, rather than to the 68 psig level of the previous run, indicating appreciably lower frictional losses. The temperature profiles in Fig. 6 indicate operational foil gradients the order of  $70^{\circ}\text{F/inch}$ .

In the absence of availability of preheated foil-lift air, the rotor was allowed to cool at 600 RPS and was coasted down when temperatures approached equilibrium. A comparison of the coastdown curves is made in Fig. 7. The first curve corresponds to a condition in which both rotor and foil temperatures remained close to ambient. The second curve reflects a slight increase in friction when coasting down after near equilibrium temperatures were attained, following cutoff of external heat input. Differences in rotor response during coastdown from near equilibrium temperatures, prior to and following the 8-hour run, were minor, and replacement of one of the four molybdenum foils by an Inconel foil had no drastic effect on the rotor response.

In the course of preliminaries to experiments at elevated temperatures, attempts were made to operate the rotor at high speeds and in the pressurized mode. The objective was to provide a relatively large and safe operating clearance during coastdown, without cooling at rated speed. Early pressurization would insure against excessive reduction of gap width at low speeds when coupled with an adverse combination of thermal expansions. (Indeed, with preheated gas readily available in a turbomachine, the full benefit of early pressurization could be realized without the conflicting effect of local cooling of the foil by orifice jets experienced during the first non-isothermal test.) The result of these preliminary experiments

was very gratifying, because stable operation in the pressurized mode and in the vertical attitude was achieved up to 36,000 RPM. The high-speed orbits were negligibly small and no additional resonances have been observed. It would appear, therefore, that high-speed operation is also possible in the pressurized mode and gap width the order of 0.002 inch.

### WIPE-WEAR CHARACTERISTICS

The "forgiving" nature of foil bearings was discussed in considerable detail in references [2,3,4]. Indeed, our past and present experiences confirm the ability of the foil bearing to accommodate geometrical imperfections and misalignment, as well as thermal distortions. Furthermore, the assortment of dust and debris "digested" by the present foil bearings without any ill effects would have sufficed to cause failure in a host of conventional gas bearings.\* Thus, regardless of surface compatibility of materials, the passage of foreign particles is facilitated by the ability of the foil to deflect and to conform locally. Finally, since closure of the gap involves the entire region of wrap, rather than a line along the minimum clearance of a rigid bearing, contact is progressive and the load more uniformly distributed among a multiplicity of asperities. Hence, wiping and burnishing takes place - especially with suitably coupled materials - rather than ploughing, galling and scoring.

The same set of foils was used throughout the entire set of experiments, with the exception of replacing the molybdenum foil adjacent to the heater by an Inconel-600 foil after the first non-isothermal test run. The foils were removed after the conclusion of tests at elevated temperatures for inspection of bearing surfaces. The state of the journal surfaces is depicted in Fig. 8, in which the upper photograph corresponds to the heated end of the rotor. The state of 12 molybdenum and of 3 Inconel-600 foil

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\*We have taken no precautions, for example, to prevent airborne particles, carried by the whirlwind of turbine-air discharge, from getting into the foil-bearing entrance zones and into the gaps. No special precautions were taken during assembly, mounting of the thermocouples and heating enclosure, or with respect to filtration of the air supply.

sectors, corresponding to the 60-degree regions of wrap, are presented in Fig. 9. The upper two rows (Inconel and molybdenum) contain foil sectors adjacent to the heater, and the last row contains sectors adjacent to the thrust bearing.

Significant surface contact occurred only in the hottest part of the bearing. The most pronounced wear track on the journal was along the outboard row of orifices. Contact pressure was due mainly to rapid contraction of the molybdenum foil along the row of cooling air jets during coastdown of the rotor, in the course of the first run. Despite rapid cooling of the molybdenum foil from approximately  $800^{\circ}\text{F}$ , the surface damage of foil segments  $A_{11}$ ,  $A_{12}$  and  $A_{13}$  was slight and their condition was hardly inferior to that of the Inconel-600 sectors  $\bar{A}_{11}$ ,  $\bar{A}_{12}$  and  $\bar{A}_{13}$ , which had not been exposed to coastdown of the rotor at elevated journal temperature. The remaining foil sectors displayed only minor wear marks, to the extent that borderlines between the regions of wrap and the adjacent foil areas were difficult to distinguish. The dark circular spots on sectors  $A_{13}$  and  $A_{23}$  correspond to locations of thermocouples on the opposite foil surface and do not represent a wear pattern.

## CONCLUSIONS

The results of experiments described in this paper confirmed that the foil bearing can operate successfully at elevated temperatures and in the presence of thermal gradients. It is concluded that detrimental effects of differential expansions and distortions can be accommodated and partly compensated by the flexibility and the extensibility of the foil. Since tension and gap width can undergo appreciable changes with temperature without adverse effects on bearing operation, it appears that the performance of foil bearings is rather insensitive to large changes of parameters; changes which could not be tolerated in other types of gas bearings.

With an approximate knowledge of temperature distribution in an actual machine and reasonable matching of coefficients of expansion of bearing elements, successful operation of foil bearings, based on the present configuration, can probably be insured without recourse to special temperature-compensation and equalization devices. To avoid local cooling and contraction of foils during coastdown, the pressurization gas can be preheated.\*

In the course of experiments, the dynamic and thermal operating conditions exceeded greatly in severity those that could be anticipated in an actual machine and bordered on virtual abuse of the foil bearings. This fact notwithstanding, examination of journal and foil surfaces revealed little damage and rather excellent wipe-wear characteristics.

The inherent stability and "forgiving" character of foil bearings indicate that this method of support may be profitably applied in the field of high-speed, high-temperature turbomachinery.

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\*By supplying the gas through circumferential, capillary slots, rather than through orifices, the gas can be automatically preheated in passing through the rotor. Preheating in a high-temperature turbomachine poses no special problem.

## ACKNOWLEDGEMENTS

This research was sponsored by the National Aeronautics and Space Administration, Lewis Research Center, Cleveland, Ohio, under contract No. NAS3-11826. The author is indebted to Mr. William J. Anderson, Bearing Branch, Fluid Systems Component Division, NASA Lewis Research Center, for his cooperation and guidance with respect to technical objectives of this investigation.

Numerous discussions with Dr. A. Eshel and Mr. M. Wildmann, both of the Ampex Corporation, contributed significantly to the success of this study. The loyal support of Mr. B. Lawson and Mr. F. Schneider is gratefully acknowledged.

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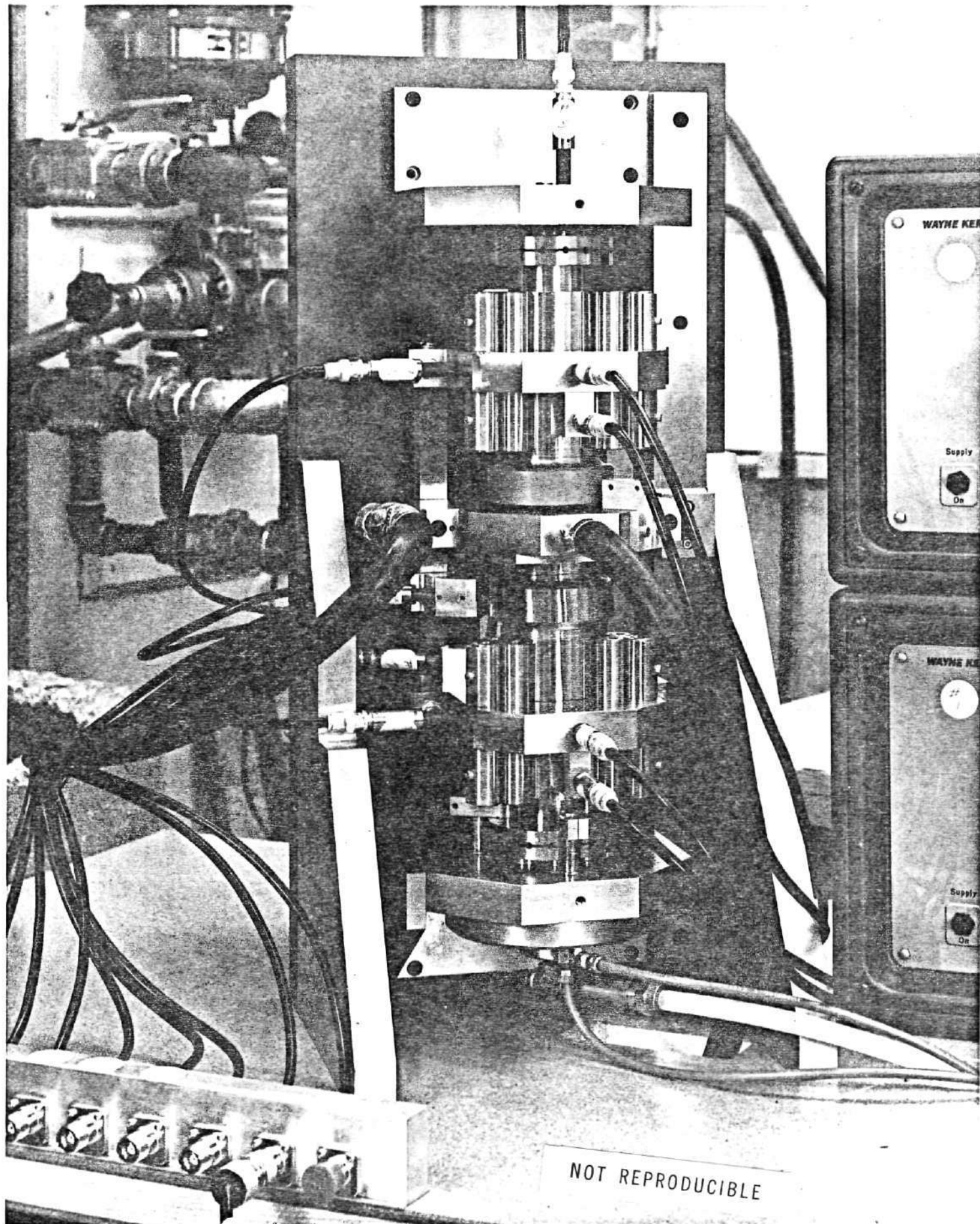


Fig. 1 View of Foil-Bearing Supported Rotor in Vertical Attitude



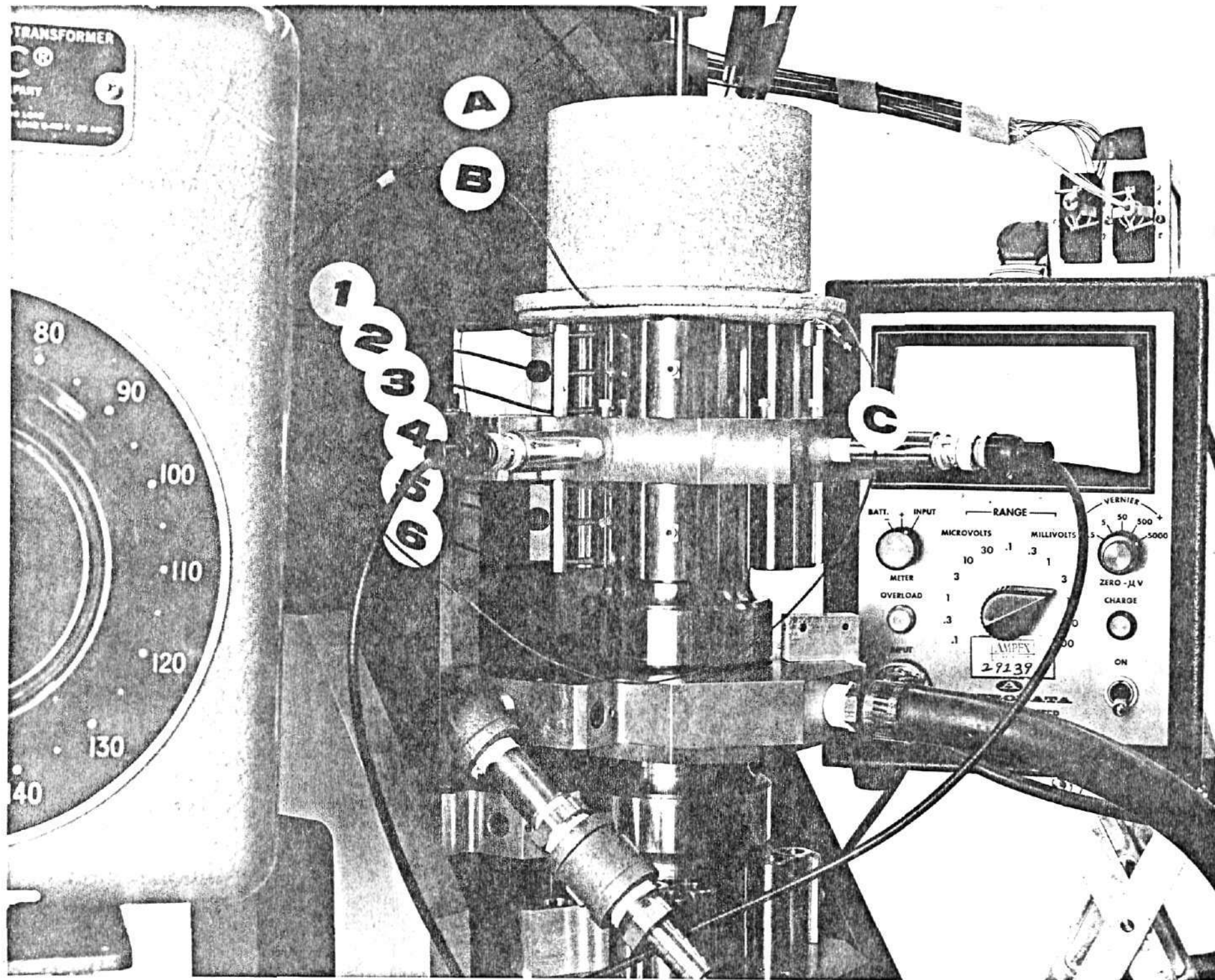


Fig. 2 View of Heater, Foil Bearing and Thermocouple Arrangement

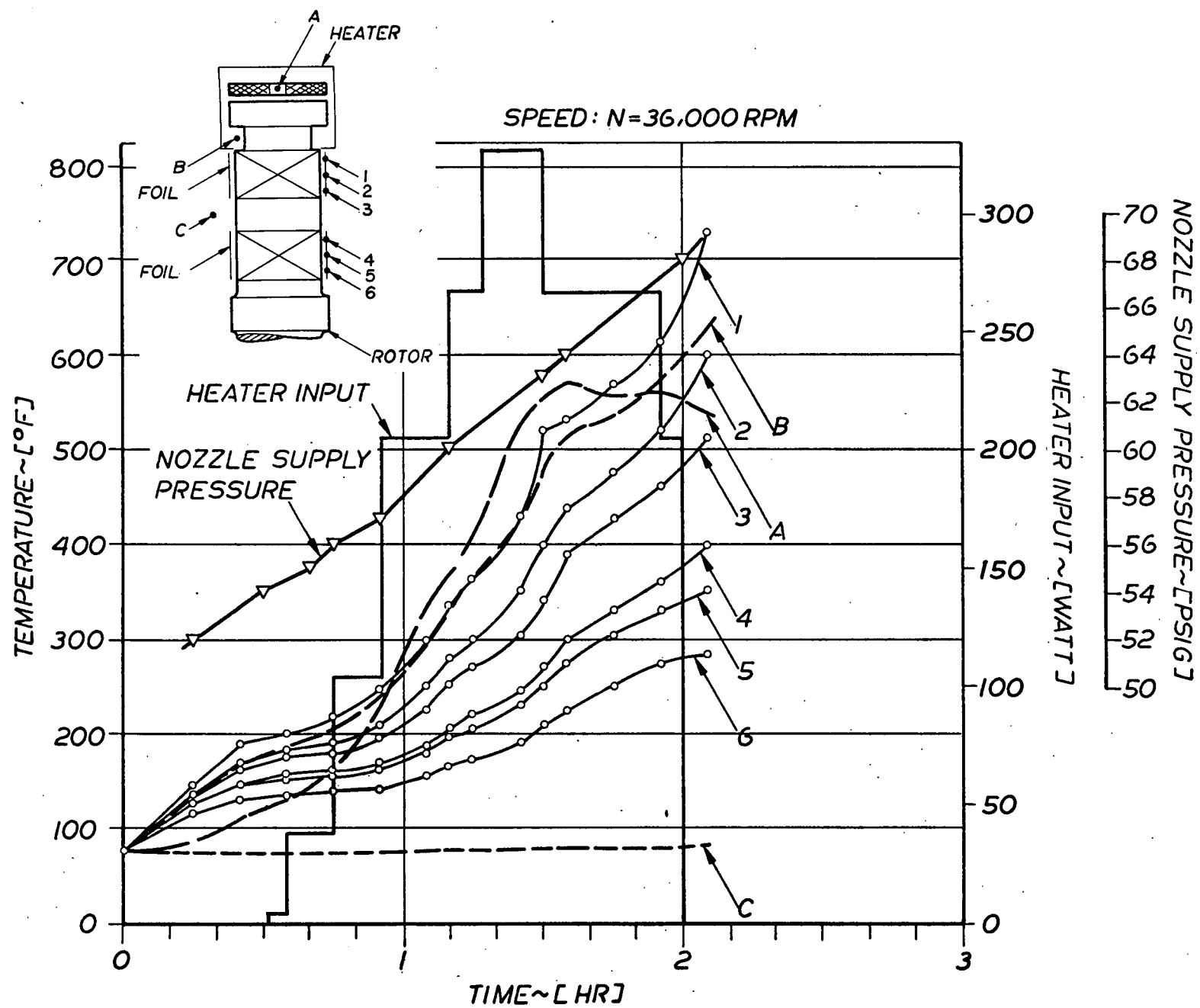


Fig. 3 Temperature Record of Heating Cycle (Rotor Vertical Molybdenum Foils)

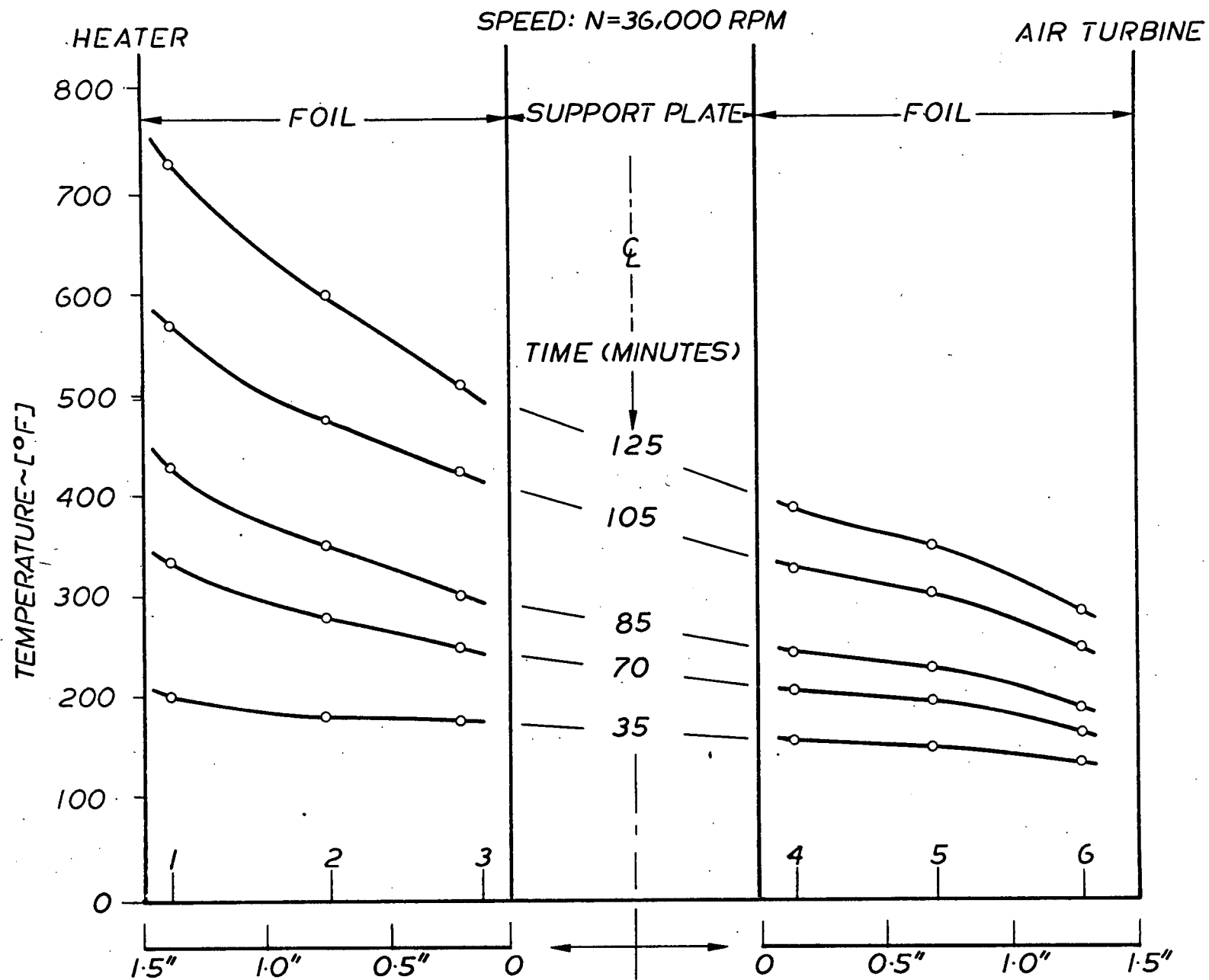


Fig. 4 Variation of Foil Temperature Along Journal Axis (Rotor Vertical  
Molybdenum Foils)

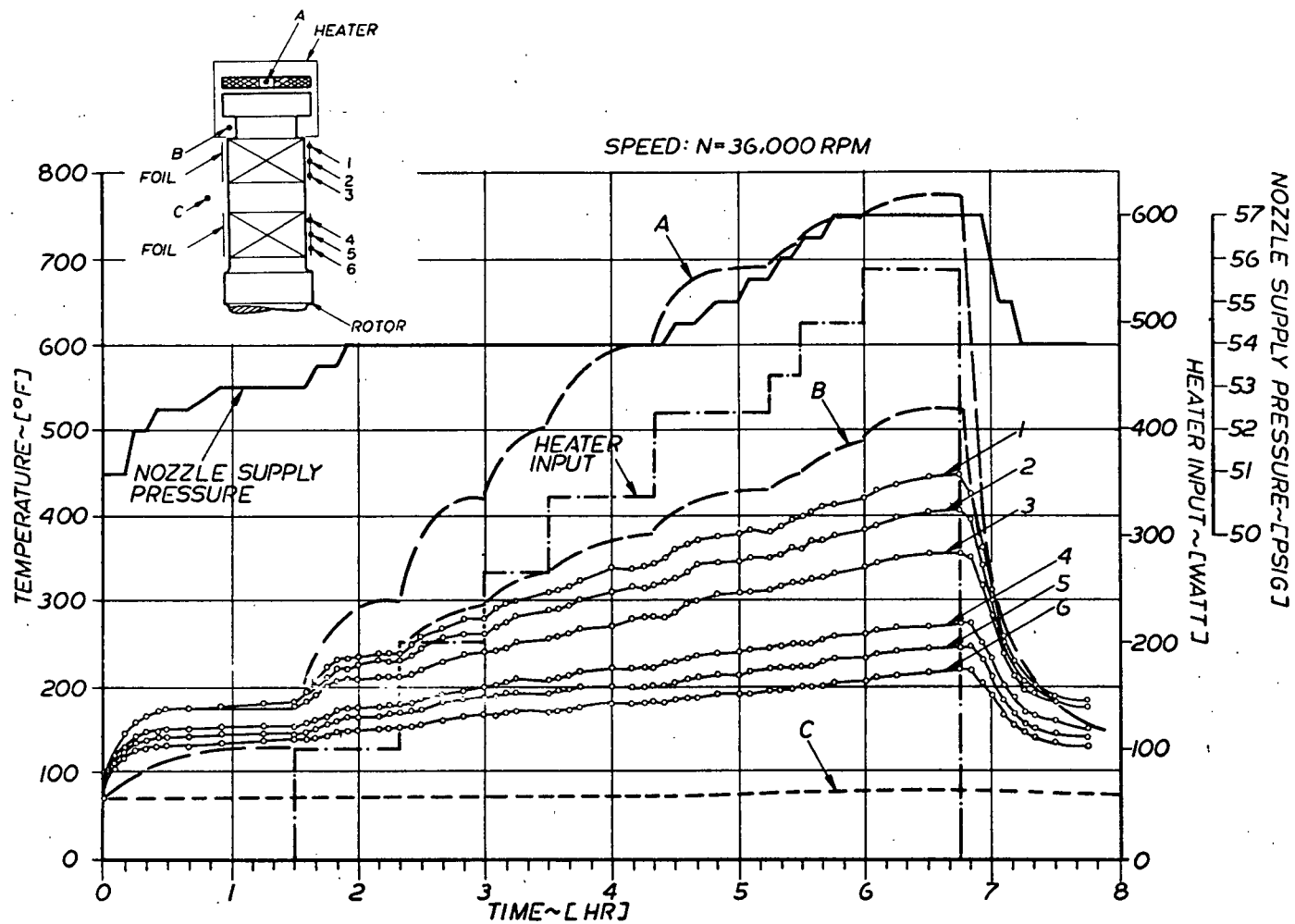


Fig. 5 Temperature Record of Heating Cycle (Rotor Vertical - Three Molybdenum and One Inconel 600 Foil)

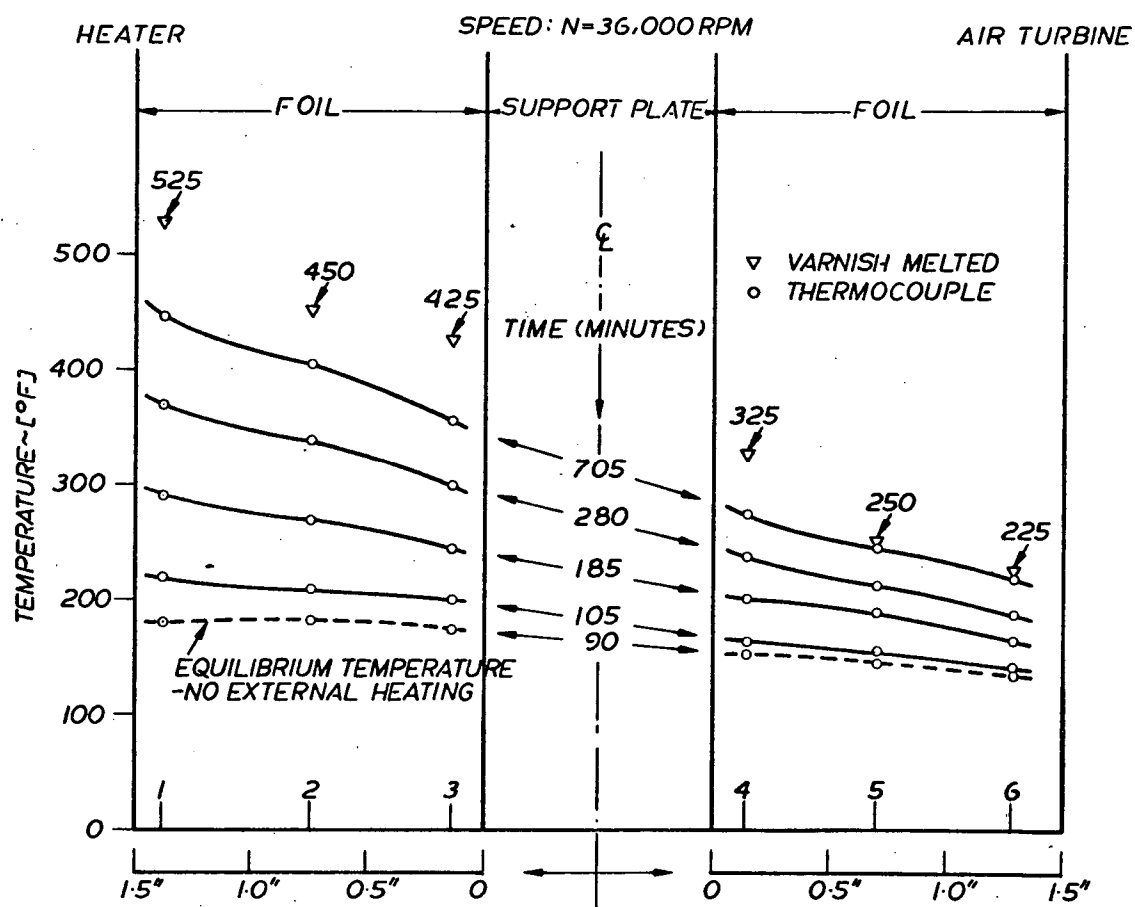


Fig. 6 Variation of Foil Temperature Along Journal Axis (Rotor Vertical - Three Molybdenum and One Inconel-600 Foil)

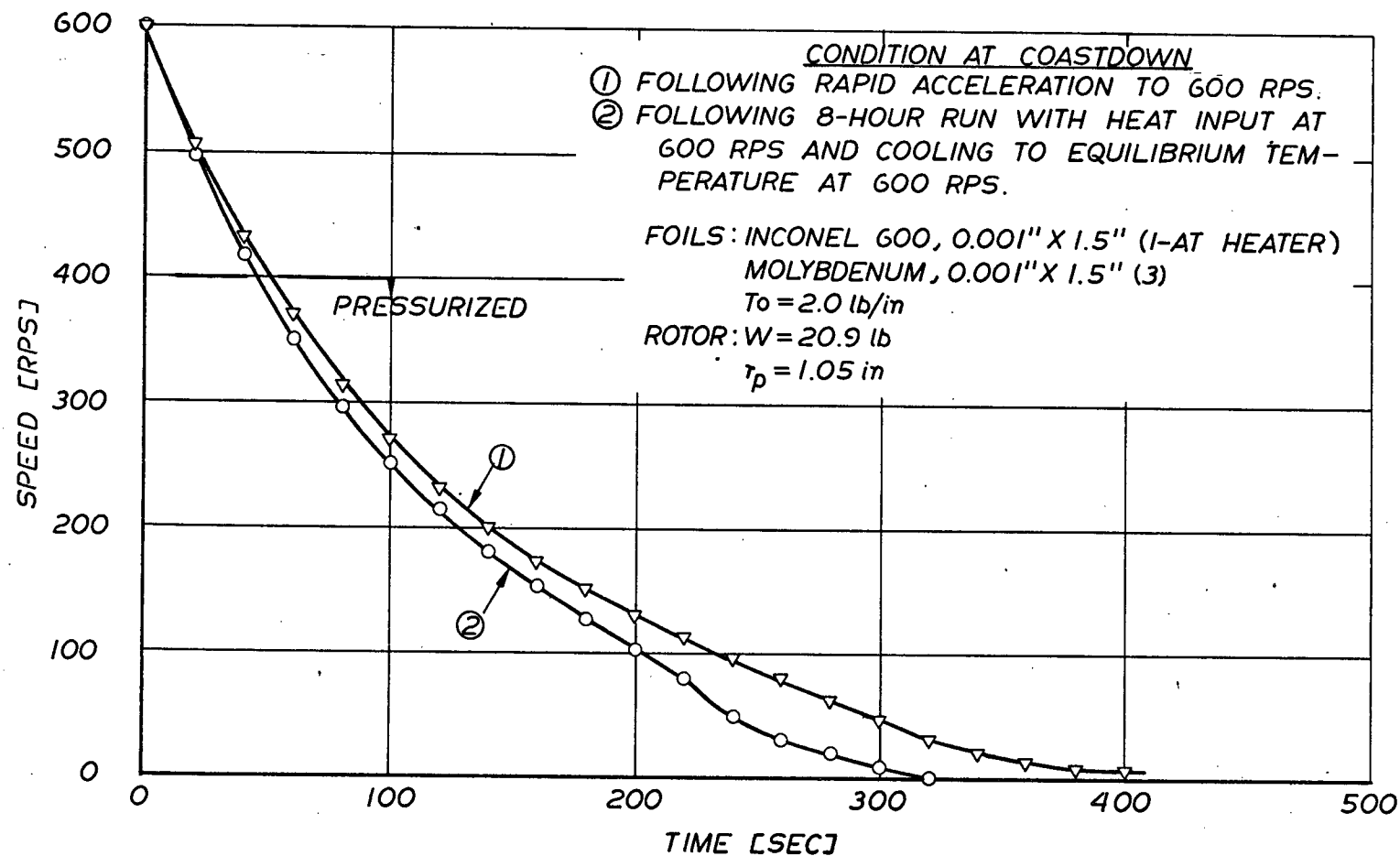
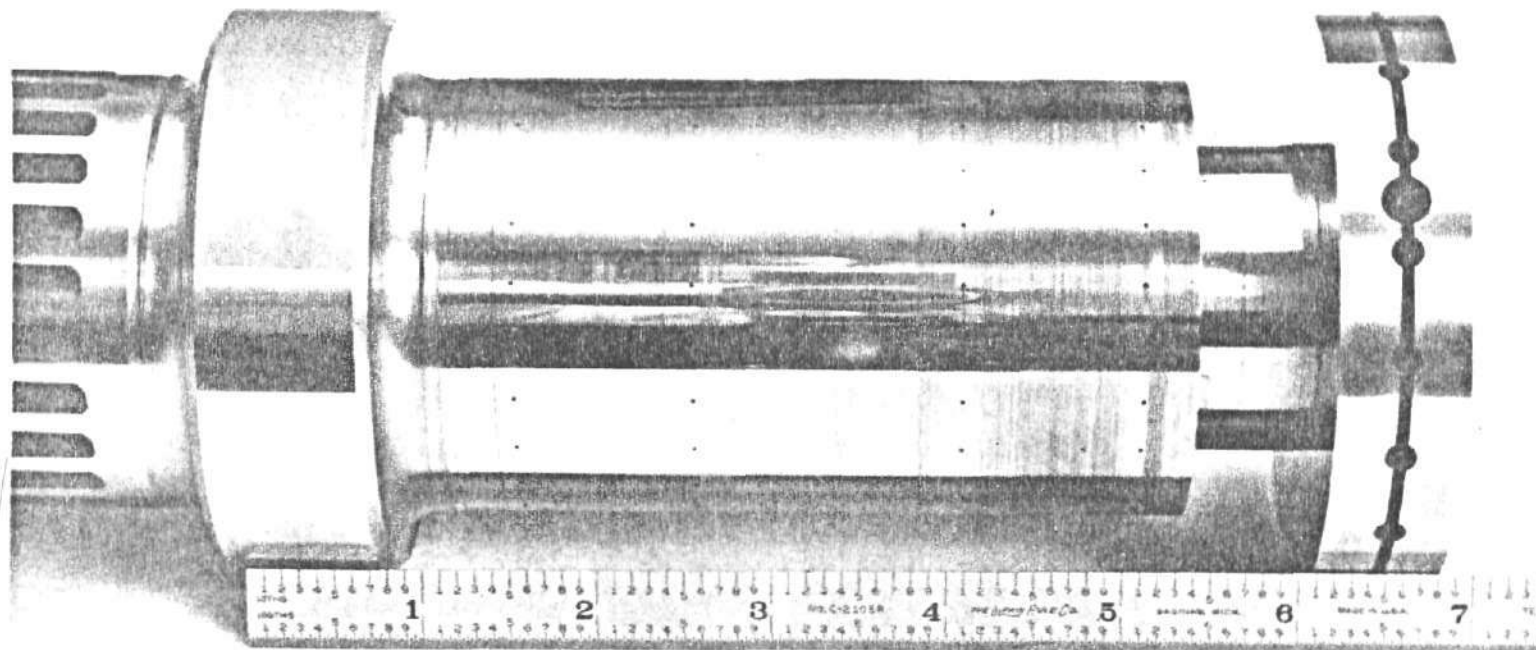
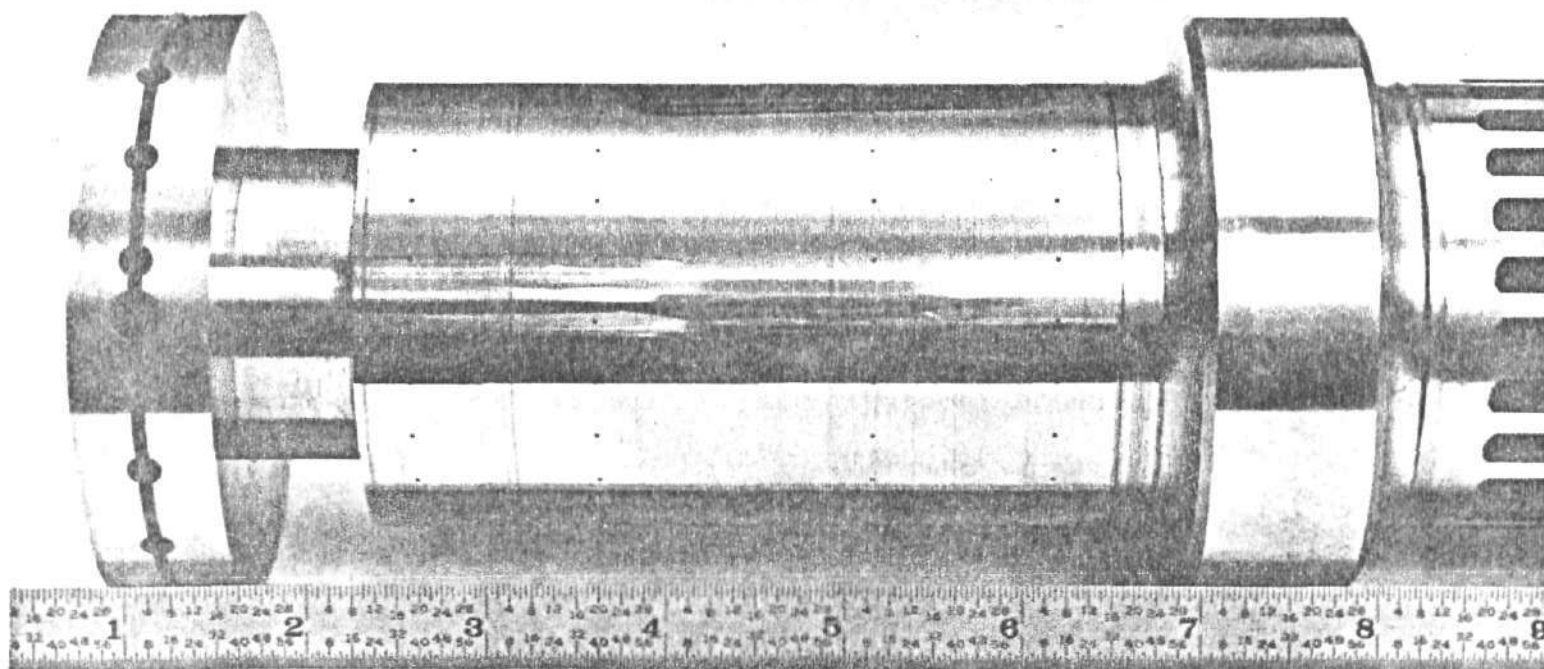


Fig. 7 Comparison of Coastdown Curves for Two Initial Conditions  
 (Rotor Vertical - Three Molybdenum and One Inconel-600 Foil)

NOT REPRODUCIBLE



a) Heated End



b) Cold End

Fig. 8 Wipe-Wear Traces on Rotor Journals

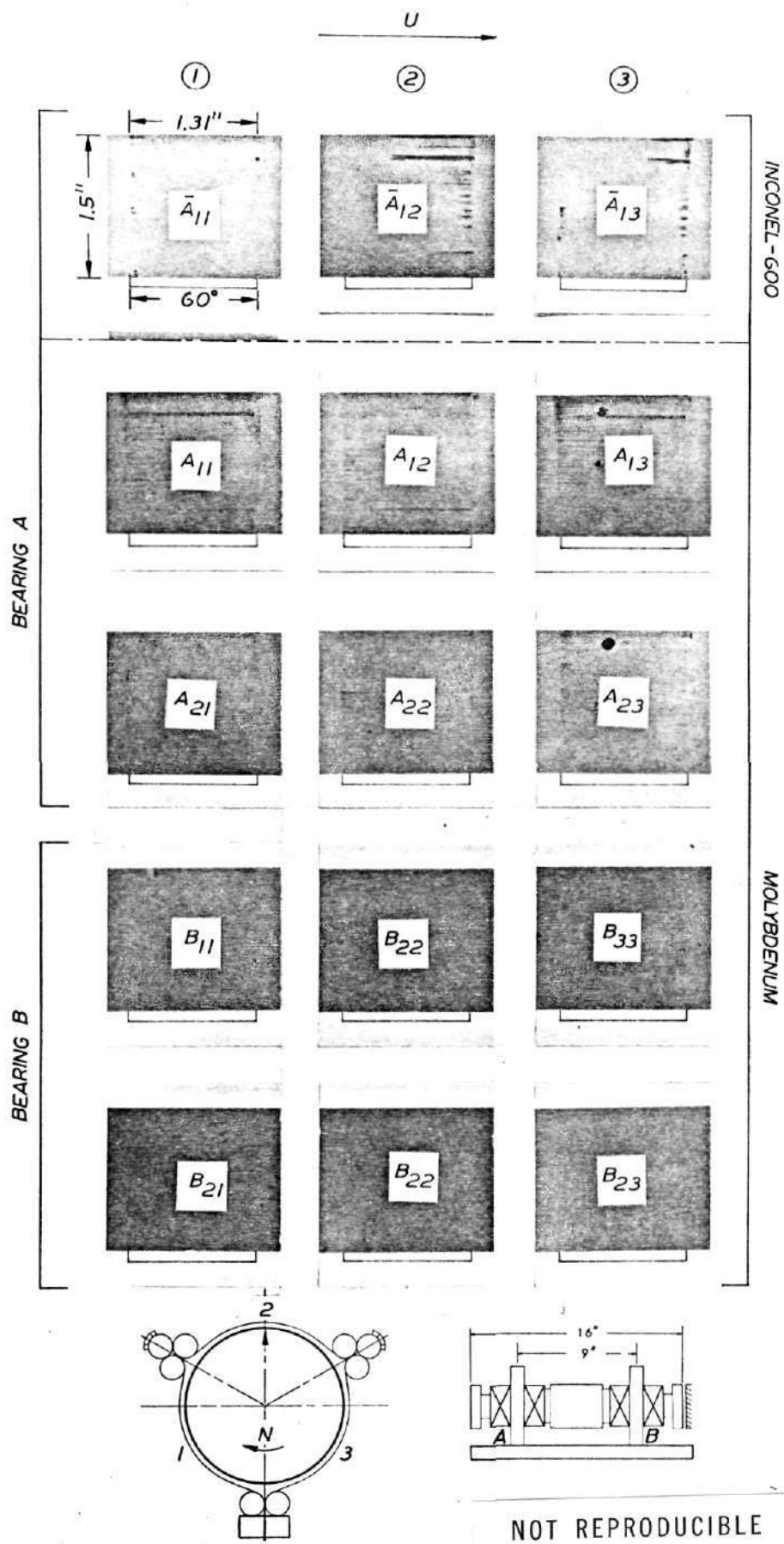


Fig. 9 Wipe-Wear Traces on Foil Sectors